'IONS FOR FLOW PROPERTY VARIATION AT OUTLET OF A CENTRIFUGAL IMPELLER

By

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Abstarct

Detailed measurements using hot-wire anemometery and high frequency Kulite transducer at centrifugal impeller outlet reveal a complex nature of flow behaviour across blade to blade pitch. The impeller outlet flow has two distinct regions called jet and wake. These regions are identified by measuring the variations of flow velocity, angle and pressure. The impeller outlet flow is characterised by defining certain non-dimensional parameters and correlating them with flow coefficient which are useful in the design of centrifugal impellers.

Nomenclature

- b impeller width (m)
- d diameter (m)
- C_p specific heat at constant pressure (kcal/kg. K)
- C absolute velocity (m/s)
- m mass flow rate (kg/s)
- P pressure (N/m^2);pressure surface
- S suction surface
- T temperature (K)
- U tip speed (m/s)
- W relative velocity (m/s)
- ϕ flow coefficient
- ρ density of the fluid (kg/m^3)
- γ ratio of specific heats
- ψ pressure rise coefficient
- β relative flow angle (deg.)
- λ mass flow fraction through wake
- ϵ ratio of wake to impeller pitch
- ν ratio of wake to jet radial velocity
- ζ ratio of wake to jet absolute flow angle

Subscripts

- j jet
- m mean
- r radial
- s static
- w wake
- ts total to static
- θ tangential
- o total condition
- 2 station at impeller outlet

Superscript

- normalised

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1.Introduction

The geometry of the centrifugal impeller is three-dimensional. The impeller channel is highly diffusive. The internal flow in the centrifugal impeller is complex due to its geometry. The impeller flow passage is rotating and has curvature both in the axial and radial directions. The flow inside the centrifugal impeller is governed by the inertia, pressure, viscous, centrifugal and Coriolis forces. These forces act on the boundary layer within the blade passages and sweep it to the suction surface shroud corner (1). Secondary flows normal to the main flow are also generated as the non-uniform inlet flow consisting of a vortex filament turns through the impeller channel (2,3). The main flow near the shroud casing wall is further influenced by the leakage flow from the pressure surface to the suction surface. As the casing wall is moving relative to the impeller the boundary layer on the casing wall is affected. The flow turbulence could stabilise or de-stabilize the flow within the impeller channel depending on the blade, hub and shroud curvatures (4,5).

The flow at impeller outlet varies with respect to space and time. The spatial variation is on blade to blade and hub to shroud planes. The measured time averaged velocity at impeller outlet using conventional pressure probes provides hub to shroud velocity distortions (6). The blade to blade circumferential distortions in velocity could be obtained either using a hot-wire anemometer or a laser velocity meter. The laser velocity meter measurements are based on statistical averaging principle and hence it cannot capture unsteady flows in centrifugal impellers(7,8). The hot-wire anemometer is highly suited to measure the unsteady flows in centrifugal impellers as they can capture instantaneous velocities (9,10,11). Flow visualisation using smoke and photographic techniques provide better understanding of flow behaviour within the impeller channel. The above measurements coupled with dynamic pressure measurements using high frequency response transducers (12,13) provide useful information about the energy transfer and its variation across the blade pitch at impeller outlet. The experience gained from measurements could be made use of to model the impeller outlet flow. One such model commonly available is called the jet-wake model (14). This paper describes the measurement and analysis of centrifugal impeller outlet flow using hot-wire anemometry and high frequency Kulite transducers. The measurements of the impeller outlet flow in the present investigation indicate a complex nature of flow behaviour across blade to blade pitch with two distinct regions called jet and wake. These regions are identified by measuring the variations of flow velocity, angle and pressure. The impeller outlet flow is characterised by defining certain non-dimensional parameters and correlating them with flow coefficients which are useful in the design of centrifugal impeller.

2.Test Facility

A centrifugal compressor of 525mm diameter, 45.5 mm width with 23 vanes backswept by 40 deg. with respect to radial direction was rotated at 5000 rpm by a D.C motor set. Thyrister control with feed back for the D.C motor ensured maintenance of the speed to an accuracy of 0.1%. The compressor and D.C motor were connected together through a step up gear-box of ratio 6. An electronic torque meter coupled in between the gear-box and the compressor was used to measure the compressor speed and power input. A bellmouth in the inlet duct was used to ensure uniform flow to the compressor. A throttle plate at the exit of the volute casing was used to vary the mass flow rate through the compressor.

3.Instrumentation

The test facility was well instrumented for detailed flow measurements at impeller outlet. A hot-wire sensor placed 8mm radially outwards from the impeller tip as shown in Fig. 1 was used in two angular positions to measure radial and whirl components of absolute velocity in a dynamic mode across the blade to blade pitch at three different locations between hub side wall and shroud side wall of the impeller. A linearizer was used in conjunction with the hot-wire anemometer circuit and calibration was carried out separately in a steady uniform flow before and after the experiments for the complete range of velocity. The linearised voltage was measured using a DC voltmeter, which had a resolution of 100 microvolts and an accuracy of +/-200 microvolts. The flow velocity was estimated from total and static pressure measurements at the calibration plane, where hot-wire was located. The pressure measurements were accurate to the order of +/-2%. The hot-wire trace captured through a computer contro dual beam signal analyser and record through its memory onto magnatic disks. Once per revolution pulse generated from an eddy current probe and shaft projection was used to trigger the hot-wire trace. The hot-wire signals were recorded for a duration of 50 milliseconds which correspond to nearly 4 revolutions of the rotor. Over this period, the signal was digitised into 4096 data recordings. This provided a frequency response of 20 kHz, while the blade passing frequency was 1.9 kHz only. Fifty such recordings one after the other in a phase locked manner were obtained to get the ensemble average of the signal, consisting of a duration to cover 92 blade passages. A general procedure was evolved to locate the suction and pressure surface of the blade trailing edge. For the measurements of pressure a high frequency miniature transducer was used to measure total and static presures in a dynamic mode at impeller outlet. The transducer signal was connected to the linear amplifier and processed through a computer controlled dual beam signal analyzer. All flow measurements were carried out through an on-line data acquisition system. Power and mass flow measurements were accurate to the order of +/- 4%.

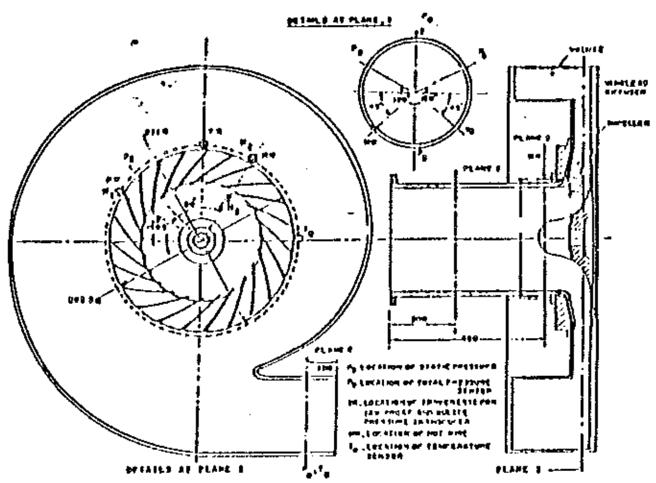

Fig. 1 Test Impeller Instrumentation

4. Results and Discussions

The detailed measurements at impeller outlet were carried out at a constant speed of 5000 rpm at the flow coefficients indicated by points A, B and C on the head-flow characteristic curve shown in Fig. 2. The time averaged

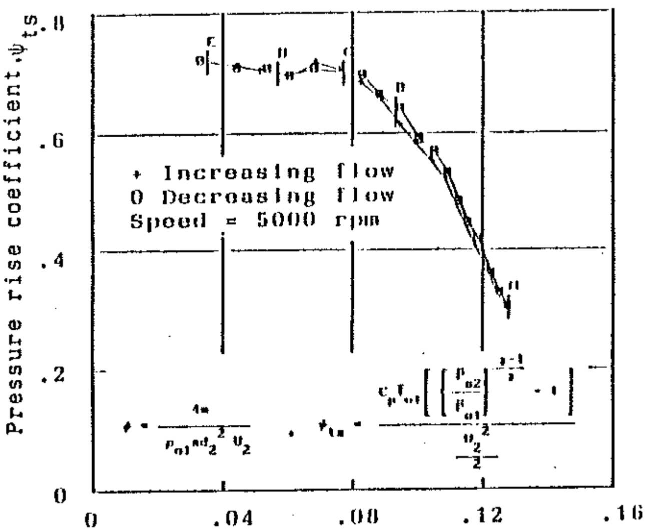

Fig. 2 Head-Flow characteristics of a centrifugal impeller

measurements were carried out using three hole yaw probe to corroborate the hot-wire measurements. The time averaged measurements show the existance of axial distortions in the radial and tangential velocities (Figs.3a-b). These flow distortions are minimum at design point. As the flow coefficient is reduced the overall blade loading increases. The increase in local blade loading with decrease in flow coefficient is larger at the hub than at the shroud.

A hot-wire probe placed 8mm radially outwards from the impeller tip was used in two angular directions to measure radial and tangential components of absolute velocity in a dynamic mode at impeller outlet. The actual velocity with which the flow left the impeller was calculated by correcting for a small amount of variation in radial location of the impeller tip point and probe point using conservation of radial and tangential momentum. The velocity signal measured by the hot-wire with respect to time was converted to velocity variation with respect to relative angular position to locate the pressure and suction surface of the blade trailing edge.

Analysis of flow structure in terms of instantaneous velocity and angle particularly in the relative frame would provide more clarity. Using the measurements of radial and tangential components of absolute velocity and impeller peripheral speed, the relative velocity and flow angle are computed.

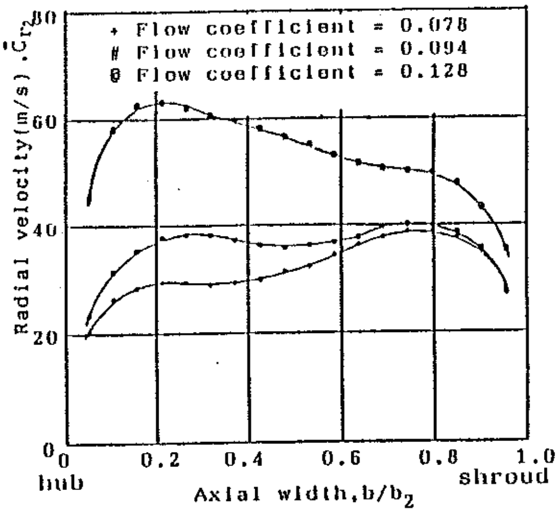

Fig. 3a Time averaged radial velocity distribution at impeller exit

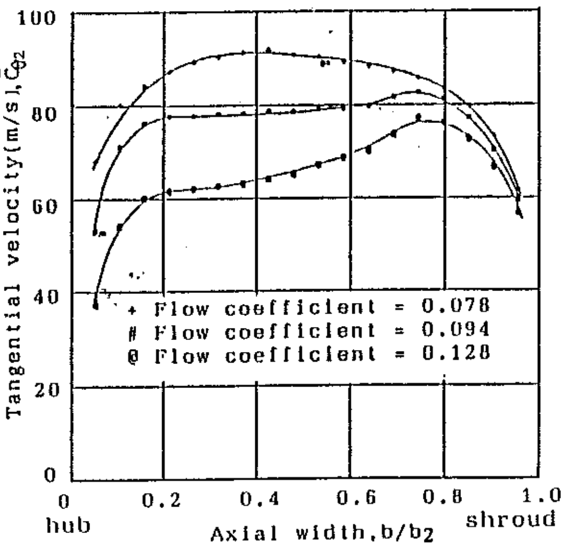

Fig. 3b Time averaged tangential velocity distribution at impeller exit

The relative velocity variation at different locations for three flow coefficients are shown in Figs. 4a to 4i. At the highest flow coefficient the relative velocity outside the blade wake remains more or less constant with a small velocity gradient across the blade pitch. The boundary layer wakes generated by the blade thickness are more prominent than the wakes developed by the high shear flow within the free stream passage. As the flow coefficient is reduced there exists a region of low relative velocity over an area near the suction surface. One can clearly identify regions of low (wake flow) and high (jet flow) relative velocity at mid channel. The width of the low relative velocity region in terms of blade pitch can be accurately assessed.

The variation of relative flow angle at different axial locations and flow coefficients are shown in Figs.5a to 5i. At the highest flow coefficient, the channel is fully dominated by jet flow and relative flow angle at the mid channel is uniform and near about the blade outlet angle. This is a typ value of outlet flow angle one would for the impeller design assuming distributed relative eddy and slip correlation. Near the side walls the flow is greatly distributed by the end wall boundary layers. As the flow coefficient is reduced, the wake flow demarked by a shear plane starts to grow slightly away from the suction surface with large variation in relative flow angle in the wake region. The relative flow angle in the jet region agrees with the value obtained from yaw probe measurements. The large variation in relative flow angle across the pitch is attributed to vortex flow generated in the shear plane within the wake region of impeller channel.

Measurements show that there is a large variation of flow velocity and angle across blade pitch. Good flow uniformity which existed at the impeller mid channel across blade pitch at high flow coefficient gets deteriorated as the flow coefficient is reduced. The total mass flow rate at impeller outlet is

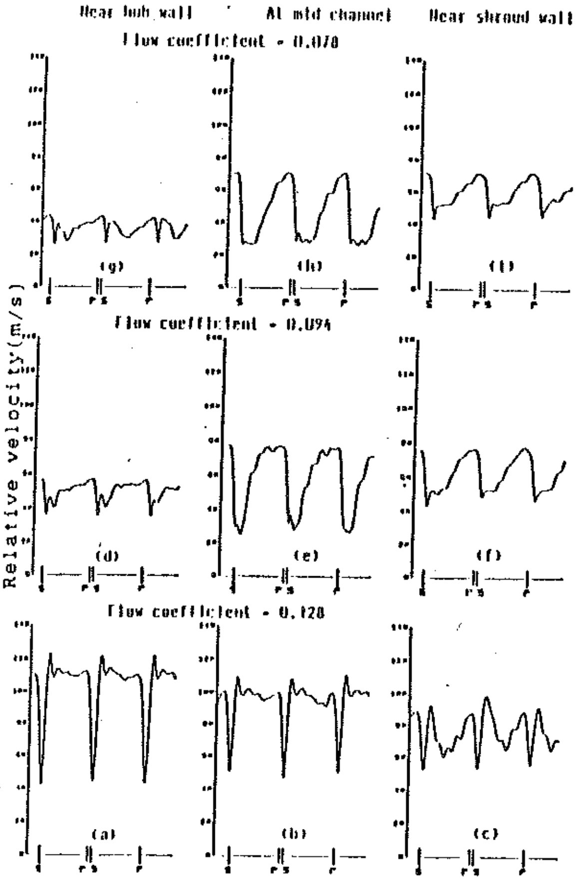

Fig. 4 Ensemble averaged instantaneous relative velocity variation at impeller exit

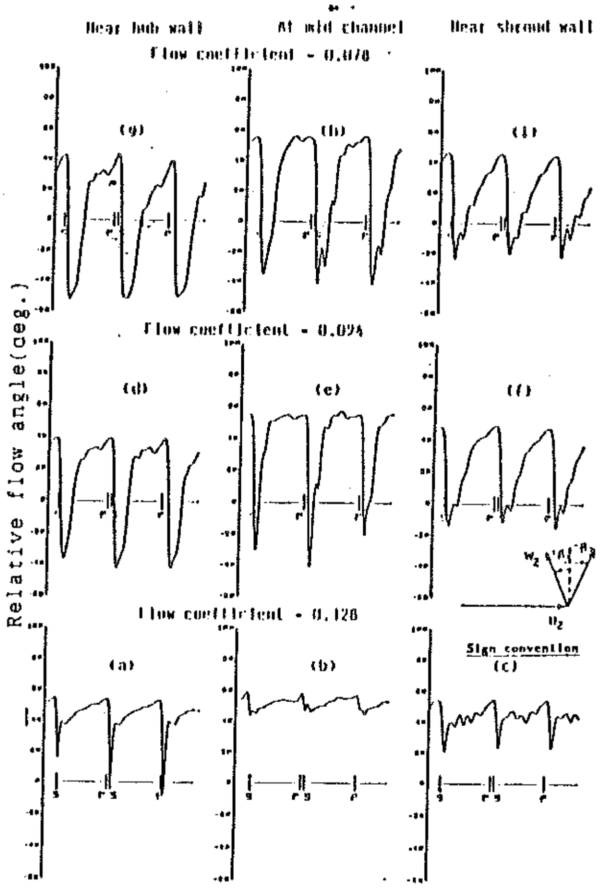

Fig. 5 Ensemble averaged instantaneous relative flow angle variation at impeller exit

between two regions with a lower ... over a region of the blade passage ... r the suction surface(wake region) and higher value in the rest (jet region) of the blade passage. The jet flow behaves like a potential flow. The measured values of velocity and angle using hot-wire anemometer in a jet region agree well with the values obtained from combination yaw probe.

High frequency response Kulite transducers were used in appropriate configurations inside a hypodermic tube to measure total pressure and static pressure variations across the blade pitch and three axial locations. Measurements show a very regular and periodic variation of total pressure at impeller outlet (Figs. 6a to 6i). A similarity of blade to blade distortion of total pressure exists at all blade passages. The total pressure in the jet flow is constant. Where as in the wake flow it can be seen to decrease linearity from a very high value close to the suction surface. Once again the growth of the wake region is seen. In the jet-wake model (14) the total pressure in the wake and jet flows are calculated from the estimated velocity and static pressure. The total pressure in the jet and wake regions from the model turns out to be different but uniform and constant.

In the jet-wake model the static pressure variation at impeller outlet is assumed to decrease linearly from pressure surface to suction surface like a potential flow distribution. The average static pressure for the jet flow is calculated from the isentropic relation. The force due to change in static pressure in the tangential direction is balanced by the centrifugal and Coriolis forces to estimate the static pressure in the wake flow. The present measurements show (Figs. 7a to 7c) that in the presence of wake flow occuring at lower flow coefficients the static pressure in the region of jet flow remains constant where as in the region of wake flow its value cannot be assessed accurately. As a first approximation it can be assumed to vary linearly from suction surface to a value of jet static pressure at the jet-wake interface. A slightly higher static pressure near the suction surface is noticed (Figs. 7g to 7i) at the highest flow coefficient. At

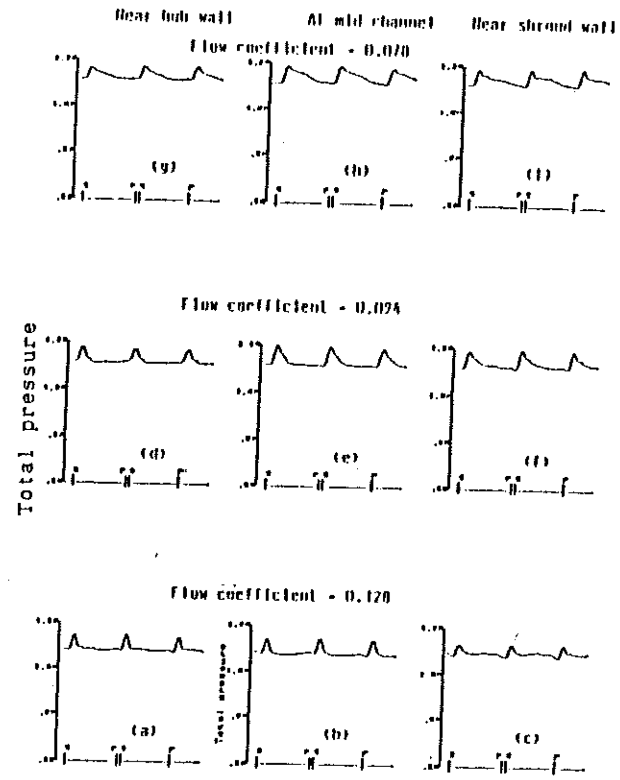

Fig. 6 Ensemble averaged instantaneous total pressure variation at impeller exit obtained from Kulite transducer

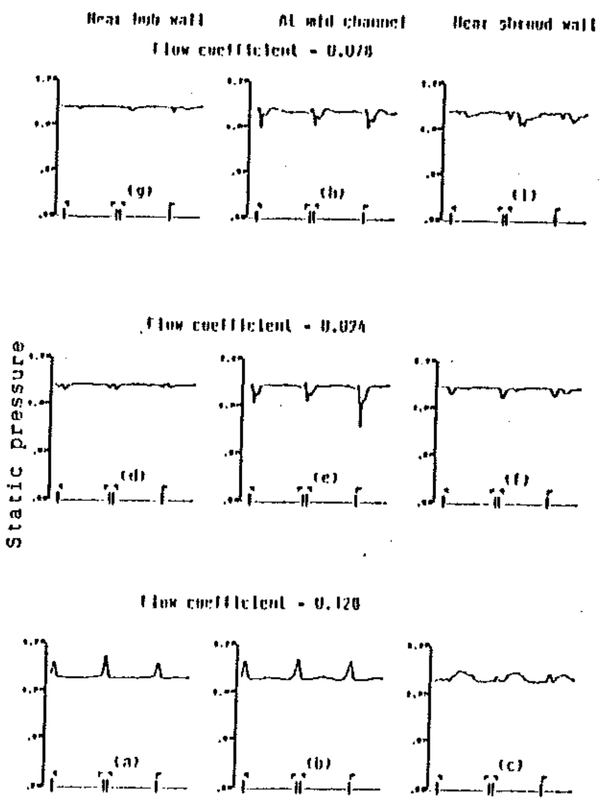

Fig. 7 Ensemble averaged instantaneous static pressure variation at impeller exit obtained from Kulite transducer

this flow coefficient the region of passage wake flow is negligible and is merged with blade wake. The rise in static pressure is attributed due to the wakes generated from the blade trailing edge thickness.

In order to check the reliability of the measured static pressure measurements the static pressure was estimated from the measured total pressure using Kulite transducer and measured absolute velocity using hot-wire anemometer both being phase locked instantaneous. Figs. 8a and 8b shows the measured and estimated static pressure variation across two blade pitch distances at the mid channel for the highest and lowest flow coefficients. It is observed from these two figures that good agreement exists both qualitatively as well as quantitatively in terms of experimental data.

___ Measured from Kulite transducer
-.- Calculated from measured total pressure

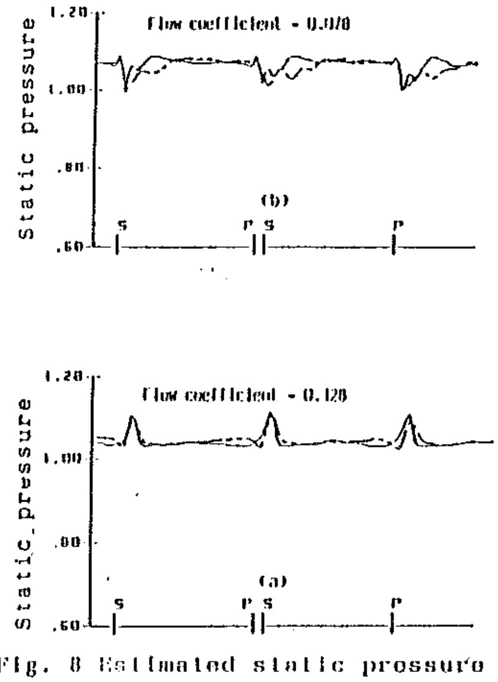

Fig. 8 Estimated static pressure at impeller exit

If one estimates the hydraulic efficiency based on the measured velocity and pressure it is observed (Figs. 9a to 9c) that the local efficiency near the suction surface and within the wake region is high. This infers that the passage wake is not a separated boundary layer with large losses. The wake flow is slightly away from the suction surface and getting energy through the fluid motion from the jet flow. This energy transfer into the wake fl dominates over the viscous losses in this region.

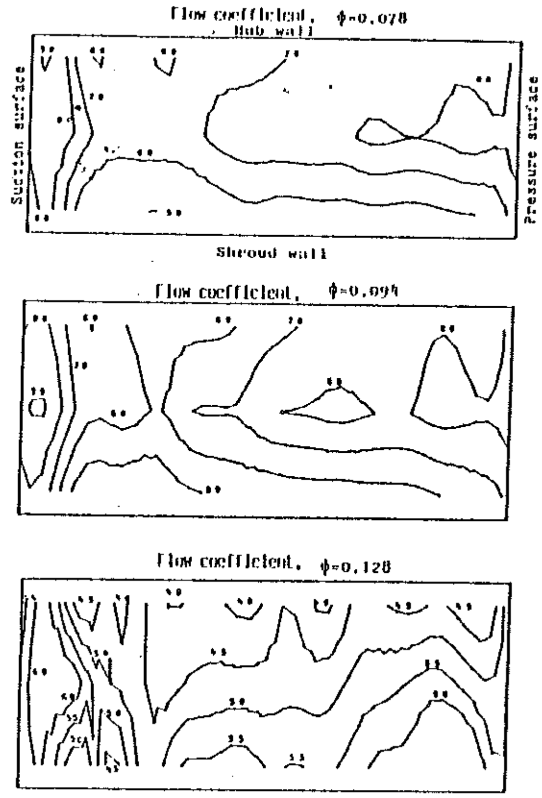

Fig. 9 Local hydraulic efficiency contours

At low flow coefficients, the flow at impeller outlet is composed of jet flow and wake region. In the jet flow the relative velocity is high and the relative flow angle is greater than the blade angle by an amount of slip. In the wake region the relative velocity is low and the relative flow angle varies drastically, wherein a constant value cannot be assigned. Based on the above findings of experimental data, the flow at impeller outlet can be characterised in terms of non-dimensional parameters such as λ, ε, ν, ζ etc., and correlated with flow coefficient. Dean (14) uses constant values for the first three parameters in the classical design of centrifugal impellers.

Empirical correlations which have been evolved for the characteristic parameters based on present experimental investigations are s flow fraction through wake

$$\lambda = 0.86 - 6.94\,\phi \quad (1)$$

Ratio of wake to impeller outlet width

$$\epsilon = 1.13 - 9.13\,\phi \quad (2)$$

Ratio of wake to jet radial velocity

$$\nu = 0.40 + 4.84\,\phi \quad (3)$$

Ratio of wake to jet absolute flow angle

$$\zeta = 1.88 + 1.04\,\phi \quad (4)$$

In the classical jet-wake model the relative flow angle in the wake is assumed equal to the blade angle itself. In the present investigation it has been found that there is a large variation of relative flow angle within the wake. An empirical relation has been fitted to describe the variation of local value of relative flow angle in the wake. It is observed from the present investigation that the relative flow angle in the wake region varies in a triangular manner with a minimum value at a distance of 1/3 from the suction surface trailing edge. This minimum value itself varies with flow coefficient and for this given impeller, the following empirical correlation is evolved.

$$\beta_{2m} = -6 + 193\,\phi - 2313\,\phi^2 + 9696\,\phi^3 \quad (5)$$

With this minimum value the variation of relative flow angle in the wake flow at any location is given by

$$\beta_{2w} = (1-3\bar{b})\,\beta_{2j} + 3\bar{b}\,\beta_{2m} \text{ for } \bar{b} \leq 1/3 \quad (6)$$

$$\beta_{2w} = [3(1-\bar{b})\,\beta_{2m} + (3\bar{b}-1)\,\beta_{2j}]/2 \quad \text{for } \bar{b} > 1/3 \quad (7)$$

Where $\bar{b} = b/\epsilon$, β_{2j} is the relative flow angle in the jet flow.

5.Conclusions

* The wake region in the freestream outlet of a centrifugal impeller passage depends on the flow coefficient and can occupy as much as 40% of the circumferencial pitch of the impeller.

* The jet-wake parameters depend on the flow coefficient particularly with back swept blading, which controls the blade loading. The measured mass flow fraction through wake and wake width depend on flow coefficient and these increase with decrease in flow coefficient.

* The location of wake flow is slightly away from the suction surface. Considerable fraction of mass flow rate goes through the wake region and the wake width itself is a large portion of the passage. Both are found to vary with flow coefficient. These variations cannot be neglected while predicting the off-design performance of the compressor.

* The empirical correlations developed here would be helpful is such prediction methods. These are borne out of experimental data and are applicable to backswept design of impellers.

6.Acknowledgements

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